

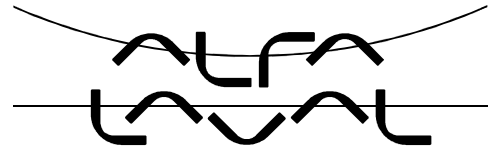
	PRESSURE VESSEL CALCULATION	Doc.No. NDC-CP21-24562-01
		Rev. 00
		Page 0/41

N° de Commande Alfa Laval / Alfa Laval Order No. CP2021-24562	Nom du Client / Customer Name DESMET
N° de Série Alfa Laval / Alfa Laval Serial No. CP40-2824	N° de Commande Client / Customer Ref No. -
Type d'Equipement Alfa Laval / Alfa Laval Equipment type CPX40 - V - 200Plates - SA 240 type 316L	Repère Client / Customer Tag No. 681B2.1 / 19-X-753B

PROJECT - WORPAR
 ITEM NBR - 681B2.1
 eB NBR - 100207389.000
 DATE - 01.09.2021

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00	20/07/21	N.ROSSIGNOL	NRL	T.SINTHOMEZ	TSZ	S.BECHU		First issue
Indice Review	Date Date	Nom Author		Verifié Checked		Approuvé Approved		Objet de la revision Review subject

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Pressure vessel calculation for COMPABLOC Heat Exchangers

I - GENERAL DATA

Pressure vessel code	ASME Boiler and Pressure Vessel Code, section VIII, Division 1 in SI units ⁽¹⁾			
Code edition	Edition 2019			
Calculation form version	ASME VIII-1.CP.19-0			
Standard version	ASME			
Heat Exchanger type	WHE_type = "CP40-V-200Plates"			
	<u>Side A</u>		<u>Side B</u>	
Design pressure	p.design = 1.04·MPa / FV		p.design. = 1.04·MPa / FV	
	p.design = 10.4·bar / FV		p.design. = 10.4·bar / FV	
	p.design = 151·psi /FV		p.design. = 151·psi /FV	
Test pressure	p..test = See Appendix 2		p..test.= See Appendix 2	
Max. design temperature	T_maxA = 120·°C	T_maxA = 248·°F	T_maxB = 120·°C	T_maxB = 248·°F
Min. design temperature	T_min = 0·°C	T_min = 32·°F	T_min. = 0·°C	T_min. = 32·°F
General drawing	Dwg_n = "DWG-CP21-24562-01"			
Material specification	ASME section II, Edition 2019 (SI units)			
Allowable stress	ASME section II, Part D, Edition 2019 (SI units)			
⁽¹⁾ SI units are used for calculation. US customary units are converted from SI units for information.				

II - Considered loads (UG 22)

<u>LOADS</u>	<u>Yes/No</u>	<u>REMARKS</u>
Internal design pressure	Yes	
External design pressure	Yes	Appendix 3 (Sheet 33 to 41)
Weight of the vessel	No	.
Weight of the contents	No	
Additional pressure due to static of liquids	Yes	
Static reaction from weight of attached equipments	No	
Internal attachments	No	
Vessel support attachment	No	.
Cyclic and dynamic reactions due to pressure	No	
Cyclic and dynamic reactions due to thermal variations	No	
Cyclic and dynamic reactions from equipment	No	
Mechanical loadings	No	
Wind reaction	No	.
Snow reaction	No	.
Seismic reaction	No	.
Impact reactions	No	
Differential thermal expansion	No	
Temperature gradients	No	
Deflagration loads	No	.
Nozzles loads calculations	Yes	FEA-CP21-24562-01
Test pressure checking with 90% yield stress	No	.

III - CALCULATION PARTS SIDE A :

1- BOLT LOADS CALCULATION. (Appendix 2)

Design pressure : $P_{\text{design}} = 1.04 \cdot \text{MPa}$ $P_{\text{design}} = 10.4 \cdot \text{bar}$

Additional pressure due to static height of liquid : $P_{\text{stat}} = 0.02 \cdot \text{MPa}$ $P_{\text{stat}} = 0.2 \cdot \text{bar}$

Calculation pressure according to UG-98 (b) : $P = P_{\text{design}} + P_{\text{stat}}$ $P_A = 1.06 \cdot \text{MPa}$ $P_A = 10.6 \cdot \text{bar}$ $P_A = 153.7 \cdot \text{psi}$

Gasket material : $\text{Mat}_{\text{gkt}} = \text{"Graphite PSM"}$

Gasket factor : $m_{\text{gkt}} = 2$ | (Characteristic gasket values are based on the
Min design seating stress : $y_{\text{gkt}} = 26 \cdot \text{MPa}$ | manufacturer's seal recommendation for the
Compabloc design and ASME table 2.5.1)

Gasket width : $N_{\text{gkt}} = 14 \cdot \text{mm}$

$b_0 = \text{Basic gasket seating width}$: $b_0 := \frac{N_{\text{gkt}}}{2} = 7 \cdot \text{mm}$ (From table 2.5.2 fig.(a))

$b = \text{effective gasket seating width} :$

$$b := \begin{cases} b_0 & \text{if } b_0 \leq 6 \cdot \text{mm} \\ C_b \cdot \sqrt{b_0 \cdot \text{mm}} & \text{otherwise} \end{cases} \quad b = 6.61438 \cdot \text{mm}$$

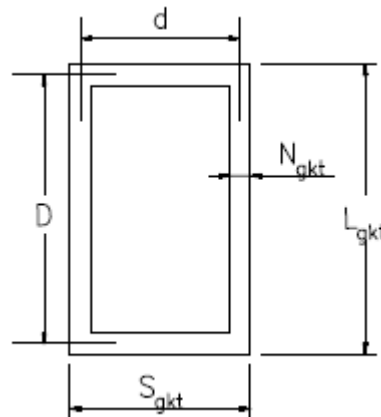
With C_b conversion factor = 2.5 for SI calculation.

Sketch of gasket :

Load reaction on line of gasket (fig 2.5.2)

Short gasket span : $S_{\text{gkt}} = 399.7 \cdot \text{mm}$
 $S_{\text{gkt}} = 15.7362 \cdot \text{in}$

Long gasket span : $L_{\text{gkt}} = 1190.7 \cdot \text{mm}$
 $L_{\text{gkt}} = 46.878 \cdot \text{in}$



Short gasket span on line : $d := S_{\text{gkt}} - (2 \cdot b) = 386.5 \cdot \text{mm}$ $d = 15.2 \cdot \text{in}$

Long gasket span on line : $D := L_{\text{gkt}} - (2 \cdot b) = 1177.5 \cdot \text{mm}$ $D = 46.4 \cdot \text{in}$

Gasket perimeter : $P_g := (d + D) \cdot 2 = 3127.9 \cdot \text{mm}$ $P_g = 123.1 \cdot \text{in}$

1.1 OPERATING LOAD

(Appendix 2 § 2.5, equation 1 adapted for rectangular gaskets)

$$W_{m1} := (D \cdot d \cdot P_A) + (2 \cdot b \cdot P_g \cdot m_{gkt} \cdot P_A) \quad W_{m1} = 570083.7 \cdot N$$

1.2 INITIAL LOAD

(Appendix 2 § 2.5, equation 2 adapted for rectangular gaskets)

$$W_{m2} := P_g \cdot b \cdot y_{gkt} \quad W_{m2} = 537914.4 \cdot N$$

2- REQUIRED BOLT AREA (Appendix 2 § 2.5(d))

Bolt material : Mat_{trod} = "SA 193 gr B7"

Allowable bolt stress :

At design temperature, Sb_{trod} = 172 · MPa

$$A_{m1} := \frac{W_{m1}}{Sb_{trod}} = 3314.4 \cdot \text{mm}^2$$

At room temperature, Sa_{trod} = 172 · MPa

$$A_{m2} := \frac{W_{m2}}{Sa_{trod}} = 3127.4 \cdot \text{mm}^2$$

Required bolt area :

$$A_m := \max(A_{m1}, A_{m2}) \quad A_m = 3314.4 \cdot \text{mm}^2$$

Actual bolt area (Ab) :

Diameter of threaded rods, d_{trod} = 24 · mm

Root area, S_{trod} = 317 · mm²

Number of threaded rods, N_{trod} = 32

Bolting area provided, Ab := N_{trod} · S_{trod} Ab = 10144 · mm²

To have adequate number and size, Ab shall be greater than 1 * A_m

(With corrective factor according mechanical strength test RGE-CP90-322)

$$1 \cdot A_m = 3314.4 \cdot \text{mm}^2 \quad (1=true, 0=false) \quad Ab > 1 \cdot A_m = 1$$

3- STUD BOLT THREAD LENGTH

According to UG-43 (g)

Nominal diameter of bolt : $d_{\text{trod}} = 24 \cdot \text{mm}$ $d_{\text{trod}} = 0.9449 \cdot \text{in}$

Allowable stress of bolt at design temperature : $S_{b_{\text{trod}}} = 172 \cdot \text{MPa}$

Allowable stress of girder at design temperature : $S_{b_{\text{girder}}} = 118 \cdot \text{MPa}$

$$L_{\text{threadmin}} := \min \left(1.5 \cdot d_{\text{trod}}, \max \left(0.75 \cdot d_{\text{trod}} \cdot \frac{S_{b_{\text{trod}}}}{S_{b_{\text{girder}}}}, d_{\text{trod}} \right) \right)$$

$L_{\text{threadmin}} = 26.2 \cdot \text{mm}$
 $L_{\text{threadmin}} = 1.033 \cdot \text{in}$

Thread length : $L_{\text{thread}} = 36 \cdot \text{mm}$ $L_{\text{thread}} = 1.417 \cdot \text{in}$

Validated criterion : $L_{\text{threadmin}} < L_{\text{thread}} = 1$
(1=true, 0=false)

4- PANEL CALCULATION

UNSTAYED FLAT HEAD CALCULATION WITHOUT OPENING: (UG 34)

Flange design bolt load calculation : (Appendix 2, § 2.5)

Under operating conditions : (equation 4)

$$W1 := W_{m1} \quad W1 = 570083.7 \cdot \text{N}$$

Under gasket seating conditions : (equation 5)

$$W2 := \frac{(A_m + A_b) S_{a_{\text{trod}}}}{2} \quad W2 = 1157425.9 \cdot \text{N}$$

Bolt load used on each thread rod:

$$F_{b.} := \frac{\max(W1, W2)}{N_{\text{trod}}} = 36170 \cdot \text{N} \quad F_{b.} = 36170 \cdot \text{N}$$



IV - CALCULATION PARTS SIDE B :

1- BOLT LOADS CALCULATION. (Appendix 2)

Design pressure : $P_{\text{design}} = 1.04 \cdot \text{MPa}$ $P_{\text{design}} = 10.4 \cdot \text{bar}$

Additional pressure due to static height of liquid : $P_{\text{stat}} = 0.02 \cdot \text{MPa}$ $P_{\text{stat}} = 0.2 \cdot \text{bar}$

Calculation pressure according to UG-98 (b) : $P = P_{\text{design}} + P_{\text{stat}}$ $P_B = 1.06 \cdot \text{MPa}$ $P_B = 10.6 \cdot \text{bar}$ $P_B = 153.7 \cdot \text{psi}$

Gasket material : $\text{Mat}_{\text{gkt}} = \text{"Graphite PSM"}$

Gasket factor : $m_{\text{gkt}} = 2$ | (Characteristic gasket values are based on the
Min design seating stress : $y_{\text{gkt}} = 26 \cdot \text{MPa}$ | manufacturer's seal recommendation for the
Compabloc design and ASME table 2.5.1)

Gasket width : $N_{\text{gkt}} = 14 \cdot \text{mm}$

b_0 = Basic gasket seating width : $b_0 := \frac{N_{\text{gkt}}}{2} = 7 \cdot \text{mm}$ (From table 2.5.2 fig.(a))

b = effective gasket seating width :

$b := \begin{cases} b_0 & \text{if } b_0 \leq 6 \cdot \text{mm} \\ C_b \cdot \sqrt{b_0 \cdot \text{mm}} & \text{otherwise} \end{cases}$ $b = 6.61438 \cdot \text{mm}$

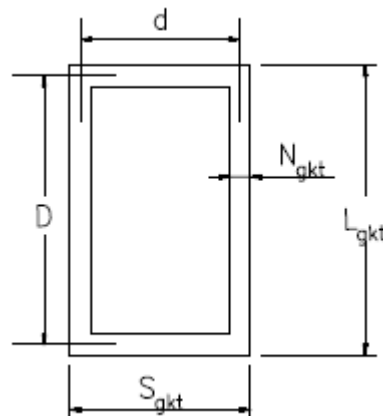
With C_b conversion factor = 2.5 for SI calculation.

Sketch of gasket :

Load reaction on line of gasket (fig 2.5.2)

Short gasket span : $S_{\text{gkt}} = 399.7 \cdot \text{mm}$
 $S_{\text{gkt}} = 15.7362 \cdot \text{in}$

Long gasket span : $L_{\text{gkt}} = 1190.7 \cdot \text{mm}$
 $L_{\text{gkt}} = 46.878 \cdot \text{in}$



Short gasket span on line : $d := S_{\text{gkt}} - (2 \cdot b) = 386.5 \cdot \text{mm}$ $d = 15.2 \cdot \text{in}$

Long gasket span on line : $D := L_{\text{gkt}} - (2 \cdot b) = 1177.5 \cdot \text{mm}$ $D = 46.4 \cdot \text{in}$

Gasket perimeter : $P_g := (d + D) \cdot 2 = 3127.9 \cdot \text{mm}$ $P_g = 123.1 \cdot \text{in}$

1.1 OPERATING LOAD

(Appendix 2 § 2.5, equation 1 adapted for rectangular gaskets)

$$W_{m1} := (D \cdot d \cdot P_B) + (2 \cdot b \cdot P_g \cdot m_{gkt} \cdot P_B) \quad W_{m1} = 570083.7 \cdot N$$

1.2 INITIAL LOAD

(Appendix 2 § 2.5, equation 2 adapted for rectangular gaskets)

$$W_{m2} := P_g \cdot b \cdot y_{gkt} \quad W_{m2} = 537914.4 \cdot N$$

2- REQUIRED BOLT AREA (Appendix 2 § 2.5(d))

Bolt material : Mat_{trod} = "SA 193 gr B7"

Allowable bolt stress :

$$\text{At design temperature, } S_{b_{trod}} = 172 \cdot \text{MPa} \quad A_{m1} := \frac{W_{m1}}{S_{b_{trod}}} = 3314.4403 \cdot \text{mm}^2$$

$$\text{At room temperature, } S_{a_{trod}} = 172 \cdot \text{MPa} \quad A_{m2} := \frac{W_{m2}}{S_{a_{trod}}} = 3127.4092 \cdot \text{mm}^2$$

Required bolt area :

$$A_m := \max(A_{m1}, A_{m2}) \quad A_m = 3314.4 \cdot \text{mm}^2$$

Actual bolt area (Ab) :

Diameter of threaded rods, d_{trod} = 24 · mm

Root area, S_{trod} = 317 · mm²

Number of threaded rods, N_{trod} = 32

Bolting area provided, Ab := N_{trod} · S_{trod} Ab = 10144 · mm²

To have adequate number and size, Ab shall be greater than 1 * A_m

(With corrective factor according mechanical strength test RGE-CP90-322)

$$1 \cdot A_m = 3314.4 \cdot \text{mm}^2 \quad (1=\text{true}, 0=\text{false}) \quad Ab > 1 \cdot A_m = 1$$

3- STUD BOLT THREAD LENGTH

According to UG-43 (g)

Nominal diameter of bolt :

$$d_{\text{trod}} = 24 \cdot \text{mm}$$

$$d_{\text{trod}} = 0.945 \cdot \text{in}$$

Allowable stress of bolt at design temperature :

$$S_{b_{\text{trod}}} = 172 \cdot \text{MPa}$$

Allowable stress of girder at design temperature :

$$S_{b_{\text{girder}}} = 118 \cdot \text{MPa}$$

$$L_{\text{threadmin}} := \min \left(1.5 \cdot d_{\text{trod}}, \max \left(0.75 \cdot d_{\text{trod}} \cdot \frac{S_{b_{\text{trod}}}}{S_{b_{\text{girder}}}}, d_{\text{trod}} \right) \right)$$

$$L_{\text{threadmin}} = 26.2 \cdot \text{mm}$$

$$L_{\text{threadmin}} = 1.033 \cdot \text{in}$$

Thread length :

$$L_{\text{thread}} = 36 \cdot \text{mm}$$

$$L_{\text{thread}} = 1.417 \cdot \text{in}$$

Validated criterion :

$$L_{\text{threadmin}} < L_{\text{thread}} = 1$$

(1=true, 0=false)

4- PANEL CALCULATION

UNSTAYED FLAT HEAD CALCULATION WITHOUT OPENING: (UG 34 equation 5)

Flange design bolt load calculation : (Appendix 2, § 2.5)

Under operating conditions : (equation 4)

$$W1b := W_{m1}$$

$$W1b = 570083.7 \cdot \text{N}$$

Under gasket seating conditions : (equation 5)

$$W2b := \frac{(A_m + A_b) S_{a_{\text{trod}}}}{2}$$

$$W2b = 1157425.9 \cdot \text{N}$$

Bolt load used on each thread rod:

$$F_b := \frac{\max(W1b, W2b)}{N_{\text{trod}}} = 36170 \cdot \text{N}$$

$$F_b = 36170 \cdot \text{N}$$



V - PANELS AND HEADS CALCULATION

Alfalaval Compabloc ® has a design that differ from these covered in ASME VIII div.1 and the following calculation way is defined :

- Use of the chapter UG-16 for these designs with the alternative of using Mandatory Appendix 46
- Use of the chapter 4.12 of Div.2 as allowed by Mandatory Appendix 46, and if the design by rule is not possible, move to next option.
- Use of part 5 of Div.2 by using a FEA calculation.

Indeed, the design of the Compabloc (panels and heads) corresponds directly to the chapter 4.12 of Div.2: “Design rules for non circular vessels”.

But in the “scope” in the paragraph 4.12.1.2 and in the “General design requirement” in the paragraph 4.12.2.8, for vessel other than described in 4.12 and for “Bolted full side or end plates...” “Part 5 shall be used”.

Moreover, for openings in kind of vessel, the Part 5 is referenced again in the paragraph 4.12.2.9 (b) : “rules for specific designs are not provided and the design shall be in accordance with Part 5”.

Specific FEA calculation : $FEA_{\text{calculation}} = \text{"FEA-CP21-24562-01"}$

This document shows that the design of those parts fulfils the requirements of the division 1 of the section VIII ASME Code with the following parameters:

1- PANELS DATA

Side A

Calculation pressure side A	:	$P_A = 1.06 \cdot \text{MPa}$	$P_A = 10.6 \cdot \text{bar}$	$P_A = 154 \cdot \text{psi}$
Design Temperature side A	:	$T_{\text{max}A} = 120 \cdot ^\circ\text{C}$		$T_{\text{max}A} = 248 \cdot ^\circ\text{F}$
Panel Aa minimal thickness	:	$Th_{Aa} = 64 \cdot \text{mm}$		$Th_{Aa} = 2.5197 \cdot \text{in}$
Panel Ab minimal thickness	:	$Th_{Ab} = 54 \cdot \text{mm}$		$Th_{Ab} = 2.126 \cdot \text{in}$

Side B

Calculation pressure side B	:	$P_B = 1.06 \cdot \text{MPa}$	$P_B = 10.6 \cdot \text{bar}$	$P_B = 154 \cdot \text{psi}$
Design Temperature side B	:	$T_{\text{max}B} = 120 \cdot ^\circ\text{C}$		$T_{\text{max}B} = 248 \cdot ^\circ\text{F}$
Panel Ba minimal thickness	:	$Th_{Ba} = 64 \cdot \text{mm}$		$Th_{Ba} = 2.5197 \cdot \text{in}$
Panel Bb minimal thickness	:	$Th_{Bb} = 54 \cdot \text{mm}$		$Th_{Bb} = 2.126 \cdot \text{in}$

2- SQUARE HEAD CALCULATION

UNSTAYED STEEL SQUARE HEAD, INSERTED AT THE EXTREMITY OF LATERAL PANELS.

Square head material : $Mat_{head} = "SA\ 516\ gr\ 60N"$

Allowable head stress (Section II part D) :

At design temperature, $S_{head} = 118 \cdot MPa$

As described above, the head calculation is done in :

Specific FEA calculation : $FEA_{calculation} = "FEA-CP21-24562-01"$

VI - LOCKINGS CALCULATION

1- STRENGTH OF LOCKING HEAD / PANEL

1.1 Data

$S_{shear} = \text{Load carried by clamp} / \text{Strength section of the square head tongues}$

Calculation pressure Max. : $P_{max} = 1.06 \cdot MPa$ $P_{max} = 10.6 \cdot bar$

Load carried by clamp : $W_{head} := d_{head}^2 \cdot P_{max} = 254506 \cdot N$

Load per link : $P_{hl} := \frac{W_{head}}{4} = 63626 \cdot N$

Mating height in compressed gasket condition : $h_{b_h} = 4 \cdot mm$ $h_{b_h} = 0.157 \cdot in$

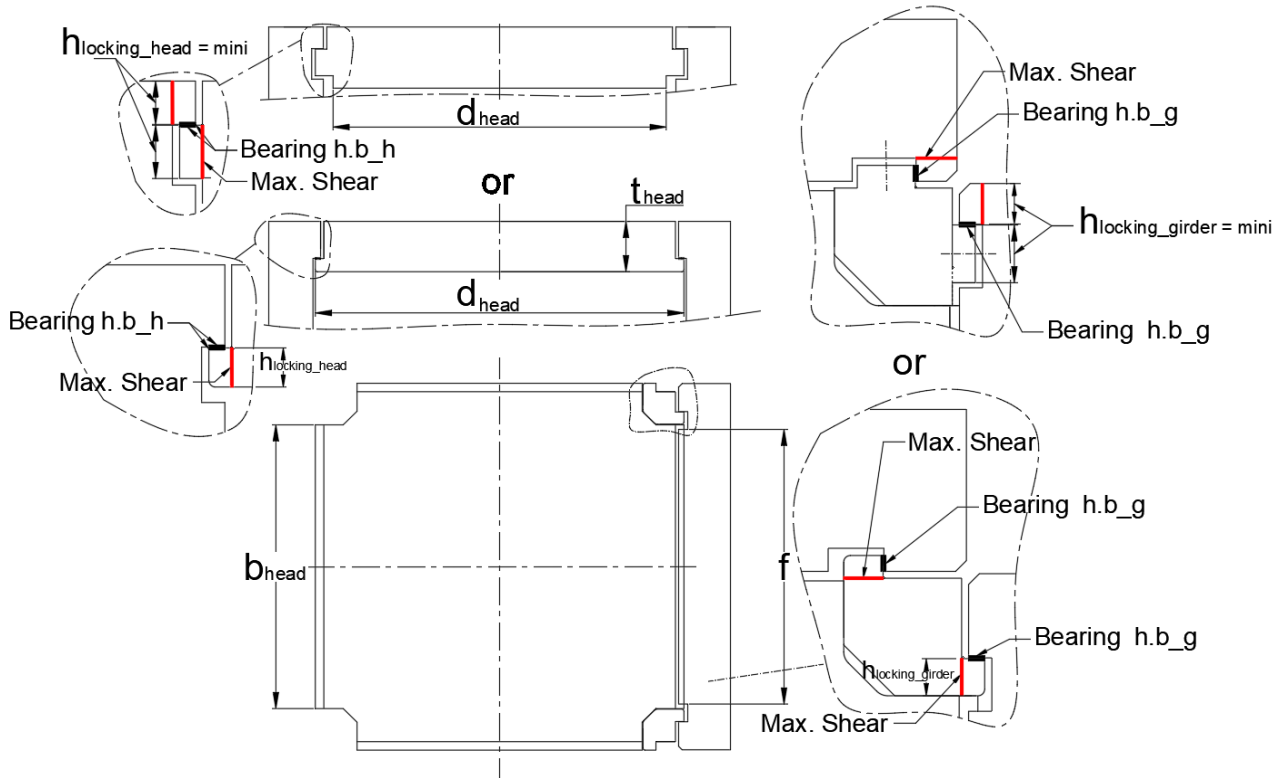
Panel locking length size : $f = 350.5 \cdot mm$ $f = 13.799 \cdot in$

Head locking length value : $b_{head} = 370 \cdot mm$ $b_{head} = 14.567 \cdot in$

Head locking height value : $h_{locking_head} = 15 \cdot mm$ $h_{locking_head} = 0.591 \cdot in$

Head square size : $d_{head} = 490 \cdot mm$ $d_{head} = 19.291 \cdot in$

Sketch of Locking :



1.2 CRITERIA DESIGN STRESS UG23 (f) and (g)

Maximum shear stress :

Strength section of the square head tongues :

$$S_{ht} := 4 \cdot h_{locking_head} \cdot \min(f, b_{head}) \quad S_{ht} = 21030 \cdot \text{mm}^2$$

Locking Shear stress :

$$S_{shear} := \frac{W_{head}}{S_{ht}} = 12.1 \cdot \text{MPa}$$

$$S_{locking_head} := \min(S_{panel}, S_{head}) \quad S_{locking_head} = 118 \cdot \text{MPa}$$

S_{shear} shall be lower than $0.8 \times S_{locking_head}$

$$0.8 \cdot S_{locking_head} = 94.4 \cdot \text{MPa} \quad S_{shear} < 0.8 \cdot S_{locking_head} = 1$$

(1=true, 0=false)

Maximum bearing stress :

$$P_{b_h} := \frac{P_{hl}}{\min(f, b_{head}) \cdot h_{b_h}} \quad P_{b_h} = 45.4 \cdot \text{MPa}$$

P_{b_h} shall be lower than $1.6 \times S_{locking_head}$

$$1.6 \cdot S_{locking_head} = 188.8 \cdot \text{MPa} \quad P_{b_h} < 1.6 \cdot S_{locking_head} = 1$$

(1=true, 0=false)

2- STRENGTH OF LOCKING GIRDER / PANEL

2.1 Data

$S_{\text{shear_girder}}$ = Load (per unit length) carried by clamp / Strength length of the girder tongues

Calculation pressure Max. :

$$P_{\text{max}} = 1.06 \cdot \text{MPa}$$

Calculation pressure Min. :

$$P_{\text{min}} = 1.06 \cdot \text{MPa}$$

Girder material :

$\text{Mat}_{\text{girder}} = \text{"SA 516 gr 60N"}$

Allowable girder stress (Section II part D) :

At max. design
temperature,

$$S_{b_{\text{girder}}} = 118 \cdot \text{MPa}$$

Square size of girder :

$$d_{\text{col}} = 59.5 \cdot \text{mm}$$

$$d_{\text{col}} = 2.343 \cdot \text{in}$$

Mating height in compressed gasket condition :

$$h_{b_g} = 4 \cdot \text{mm}$$

$$h_{b_g} = 0.157 \cdot \text{in}$$

Sketch of girder:

with : Girder locking length value
:

$$h_{\text{locking_girder}} = 15 \cdot \text{mm}$$

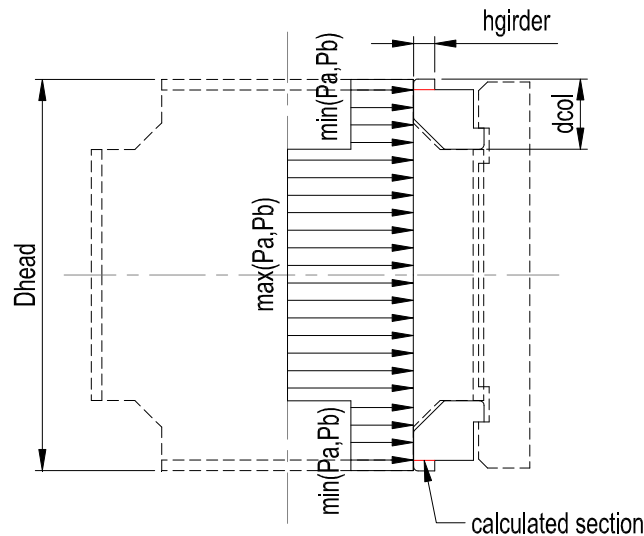
$$h_{\text{locking_girder}} = 0.591 \cdot \text{in}$$

$$D_{\text{head}} = 490 \cdot \text{mm}$$

$$D_{\text{head}} = 19.291 \cdot \text{in}$$

$$d_{\text{col}} = 59.5 \cdot \text{mm}$$

$$d_{\text{col}} = 2.343 \cdot \text{in}$$



2.2 CRITERIA DESIGN STRESS UG23 (f) and (g)

Load (per unit length) carried by clamp :

$$W_{\text{girder}} := (D_{\text{head}} - 2 \cdot d_{\text{col}}) \cdot \max(P_A, P_B) + 2 \cdot d_{\text{col}} \cdot \min(P_A, P_B)$$

$$W_{\text{girder}} = 519.4 \cdot \frac{\text{N}}{\text{mm}}$$

Strength length of the girder tongues : $S_{\text{gt}} := 2 \cdot h_{\text{locking_girder}}$

$$S_{\text{gt}} = 30 \cdot \text{mm}$$

$$S_{\text{gt}} = 1.181 \cdot \text{in}$$

$$S_{\text{locking_girder}} := \min(S_{\text{panel}}, S_{\text{b_girder}})$$

Maximum shear stress :

$$S_{\text{shear_girder}} := \frac{W_{\text{girder}}}{S_{\text{gt}}} = 17.3 \cdot \text{MPa}$$

$S_{\text{shear_girder}}$ shall be lower than $0.8 \times S_{\text{locking_girder}}$

$$S_{\text{locking_girder}} = 118 \cdot \text{MPa}$$

$$0.8 \cdot S_{\text{locking_girder}} = 94.4 \cdot \text{MPa}$$

$$S_{\text{shear_girder}} < 0.8 \cdot S_{\text{locking_girder}} = 1$$

(1=true, 0=false)

Maximum bearing stress :

$$P_{\text{b_g}} := \frac{W_{\text{girder}}}{2 \cdot h_{\text{b_g}}}$$

$$P_{\text{b_g}} = 64.9 \cdot \text{MPa}$$

$P_{\text{b_g}}$ shall be lower than $1.6 \times S_{\text{locking_girder}}$

$$1.6 \cdot S_{\text{locking_girder}} = 188.8 \cdot \text{MPa}$$

$$P_{\text{b_g}} < 1.6 \cdot S_{\text{locking_girder}} = 1$$

(1=true, 0=false)

3- MAXIMUM BOLT SPACING: (B_{smax}) according to App 2-5 (d) equation (3)

Only applicable for Lethal service

with :	Nominal bolt diameter	:	$d_{trod} = 24 \cdot \text{mm}$	$d_{trod} = 0.9449 \cdot \text{in}$
	Panel thickness (Machined at min value)	:	$Th_{Aa} = 64 \cdot \text{mm}$	$Th_{Ab} = 54 \cdot \text{mm}$
			$Th_{Ba} = 64 \cdot \text{mm}$	$Th_{Bb} = 54 \cdot \text{mm}$
	Gasket factor side A & B	:	$m_{gkt.A} = 2$	$m_{gkt.B} = 2$
	Head bolt spacing	:	$B_{s.head} = 110 \cdot \text{mm}$	$B_{s.head} = 4.3307 \cdot \text{in}$
	Column bolt spacing	:	$B_{s.col} = 98.32 \cdot \text{mm}$	$B_{s.col} = 3.8709 \cdot \text{in}$
	Column/head bolt spacing :		$B_{s.c.h} = 116.8 \cdot \text{mm}$	$B_{s.c.h} = 4.6 \cdot \text{in}$

$$B_{smax} := 2 \cdot d_{trod} + \frac{6 \cdot \min(Th_{Aa}, Th_{Ab}, Th_{Ba}, Th_{Bb})}{\max(m_{gkt.A}, m_{gkt.B}) + 0.5}$$

$B_{smax} = 177.6 \cdot \text{mm}$

The maximum head bolt spacing $B_{s.head}$, the maximum column bolt spacing $B_{s.col}$ and the column/head bolt spacing $B_{s.c.h}$ shall not exceed B_{smax}

(1=true, 0=false)
 $B_{smax} > B_{s.head} = 1$
 $B_{smax} > B_{s.col} = 1$
 $B_{smax} > B_{s.c.h} = 1$



VII - PIPE CHECKING :

Calculation pipe checking is done on the paragraph "Analytical calculation for nozzle" :

See Specific FEA calculation : FEA_{calculation} = "FEA-CP21-24562-01"

VIII - WELDING NECK AND BLIND FLANGE CHECKING :

The Welding neck flange or Long Welding neck flange must be define according to the Pressure-Temperature ratings for groups materials :

- ASME B16.5 Table 2 Ed.2013 (Nominal diameter <= DN600 (24"))
- ASME B16.47a Table 2 Ed.2017 (Nominal diameter > DN600 (24"))

Check of flange design

Side A:

Design temperature : $T_{\max A} = 120 \cdot ^\circ\text{C}$

Calculation pressure : $P_A = 1.06 \cdot \text{MPa}$ $P_A = 10.6 \cdot \text{bar}$

Side B:

$T_{\max B} = 120 \cdot ^\circ\text{C}$

$P_B = 1.06 \cdot \text{MPa}$ $P_B = 10.6 \cdot \text{bar}$

PRESSURE TEMPERATURE RATING

ASME B16.5 Ed.2013									
Table 2-1.1 PressureTemperature Ratings for Group 1.1 Materials					Table 2-2.3 PressureTemperature Ratings for Group 2.3				
Working Pressure by Classes, bar					Working Pressure by Classes, bar				
Class					Class				
Temp., °C	150	300	400	600	Temp., °C	150	300	400	600
-29 to 38	19.6	51.1	68.1	102.1	-29 to 38	15.9	41.4	55.2	82.7
50	19.2	50.1	66.8	100.2	50	15.3	40	53.4	80
100	17.7	46.6	62.1	93.2	100	13.3	34.8	46.4	69.6
150	15.8	45.1	60.1	90.2	150	12	31.4	41.9	62.8
200	13.8	43.8	58.4	87.6	200	11.2	29.2	38.9	58.3
250	12.1	41.9	55.9	83.9	250	10.5	27.5	36.6	54.9
300	10.2	39.8	53.1	79.6	300	10	26.1	34.8	52.1
325	9.3	38.7	51.6	77.4	325	9.3	25.5	34	51
350	8.4	37.6	50.1	75.1	350	8.4	25.1	33.4	50.1
375	7.4	36.4	48.5	72.7	375	7.4	24.8	33	49.5
400	6.5	34.7	46.3	69.4	400	6.5	24.3	32.4	48.6
425	5.5	28.8	38.4	57.5	425	5.5	23.9	31.8	47.7
450	4.6	23	30.7	46	450	4.6	23.4	31.2	46.8
475	3.7	17.4	23.2	34.9	A182F316L				
500	2.8	11.8	15.7	23.5					
538	1.4	5.9	7.9	11.8					
A350 Gr. LF2									



VII-NOZZLE CALCULATION

Item_{opening} = "A1-A2-B1-B2"**A- GENERAL DATA**

Acc. to section VIII Div.2. Ed 2019

Connection standard (DN - PN)	:	Dim _{nozzle} = "DN150 C1150"	
Calculation pressure	:	$P = 1.06 \cdot \text{MPa}$	$P = 10.6 \cdot \text{bar}$ $P = 153.7 \cdot \text{psi}$
Design temperature	:	$T_{\text{max}} = 120 \cdot ^\circ\text{C}$	$T_{\text{max}} = 248 \cdot ^\circ\text{F}$
Axial load at the flange	:	$F_A = 776 \cdot \text{N}$	$F_A = 174 \cdot \text{lbf}$
Moment load at the flange :		$M_y = 750 \cdot \text{N} \cdot \text{m}$	$M_y = 6638 \cdot \text{lbf} \cdot \text{in}$
		$M_z = 750 \cdot \text{N} \cdot \text{m}$	$M_z = 6638 \cdot \text{lbf} \cdot \text{in}$
Bending moment on the flange	:	$M_E = 1061 \cdot \text{N} \cdot \text{m}$	$M_E := \sqrt{M_y^2 + M_z^2} = 9388 \cdot \text{lbf} \cdot \text{in}$
Outside diameter of flange	:	$A = 280 \cdot \text{mm}$	$A = 11.0236 \cdot \text{in}$
Inside diameter of flange	:	$B = 155.1 \cdot \text{mm}$	$B = 6.1063 \cdot \text{in}$
Bolt circle diameter	:	$C = 241.3 \cdot \text{mm}$	$C = 9.5 \cdot \text{in}$
Thick. of hub at back of flange	:	$g_1 = 18.45 \cdot \text{mm}$	$g_1 = 0.7264 \cdot \text{in}$
Minimum Thick. of hub flange	:	$g_0 = 6.6 \cdot \text{mm}$	$g_0 = 0.2598 \cdot \text{in}$
Real hub length	:	$h_r = 52.8 \cdot \text{mm}$	$h_r = 2.0787 \cdot \text{in}$
Small thickness of hub (pipe)	:	$g_0 = 6.6 \cdot \text{mm}$	$g_0 = 0.2598 \cdot \text{in}$
Flange thickness in corroded condition	:	$t = 23.9 \cdot \text{mm}$	$t = 0.9409 \cdot \text{in}$

Geometrical validation :

Integral Type Flanges with a Hub and a weld (geometric criteria Figure 4.16.2)

$$\text{Slope} := \frac{(g_1 - g_0)}{h_r} \quad \text{Slope} = 0.2244 \quad \text{Slope} < \frac{1}{3} = 1$$

If slope < 1/3: the Figure 4.16.2 (a) is used without more condition

If slope > 1/3: the Figure 4.16.2 (b) is used with the following condition

Length of tube part of flange

$$h_2 = 10.3 \cdot \text{mm}$$

Minimal half of gap for the weld

$$\text{gap} = 0.5 \cdot \text{mm}$$

$$1.5g_0 = 9.9 \cdot \text{mm}$$

$$h_2 + \text{gap} \geq 1.5g_0 = 1$$

B- BOLT LOADS CALCULATION. (§ 4.16.6)

Calculation pressure : $P = 1.06 \cdot \text{MPa}$

External pressure : $P_{\text{ext}} := 0.103 \text{MPa}$

Considered gasket : $\text{Mat}_{\text{gkt}} = \text{"Graphite PSM"}$

Gasket factor : $m_{\text{gkt}} = 2$

Min design seating stress : $y_{\text{gkt}} = 26 \cdot \text{MPa}$ (From table 4.16.1 or gasket manufacturers)

Sketch of gasket :

Gasket outer diameter : $D_{\text{gkt}} = 215.9 \cdot \text{mm}$

Gasket inner diameter : $d_{\text{gkt}} = 168 \cdot \text{mm}$

Gasket width : $N_{\text{gkt}} = 24 \cdot \text{mm}$

b_0 = Basic gasket seating width:

(From table 4.16.3 fig.(1a))

$$b_0 := \frac{N_{\text{gkt}}}{2} \quad b_0 = 12 \cdot \text{mm}$$

b = effective gasket seating width :

$C_{\text{ul}} := 25.4$ conversion factor for SI calculation.

$$b := \begin{cases} b_0 & \text{if } b_0 \leq 6 \cdot \text{mm} \\ 0.5 \cdot C_{\text{ul}} \cdot \sqrt{\frac{b_0}{C_{\text{ul}}}} \cdot \text{mm} & \text{otherwise} \end{cases} \quad b = 8.7 \cdot \text{mm}$$

Gasket load reaction diameter:

$$G := D_{\text{gkt}} - (2 \cdot b) \quad G = 198.5 \cdot \text{mm}$$

Operating load

$$W_o := 0.785 \cdot G^2 \cdot P + 2 \cdot b \cdot \pi \cdot G \cdot m_{\text{gkt}} \cdot P \quad W_o = 55825.5 \cdot \text{N}$$

$$W_{\text{oext}} := 0.785 \cdot G^2 \cdot P_{\text{ext}} + 2 \cdot b \cdot \pi \cdot G \cdot m_{\text{gkt}} \cdot P_{\text{ext}} \quad W_{\text{oext}} = 5424.6 \cdot \text{N}$$

Initial load

$$W_{\text{gs}} := \pi \cdot b \cdot G \cdot y_{\text{gkt}} = 31778.5224 \cdot \text{lbf} \quad W_{\text{gs}} = 141357.9 \cdot \text{N}$$

Required bolt Area

Bolt material : $\text{Mat}_{\text{trod}} = \text{"SA 193 gr B7"}$

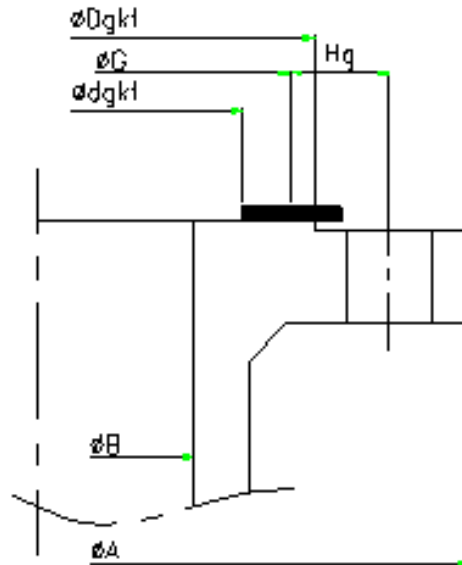
Allowable bolt stress :

At room temperature, $S_{\text{bg}} = 172 \cdot \text{MPa}$

At design temprature, $S_{\text{bo}} = 172 \cdot \text{MPa}$

Total required bolt area:

$$A_m := \max \left(\frac{W_o + F_A + \frac{4 \cdot M_E}{G}}{S_{\text{bo}}}, \frac{W_{\text{gs}}}{S_{\text{bg}}} \right) \quad A_m = 821.8 \cdot \text{mm}^2$$



Actual bolt area (A_b) :

Diameter of bolts, $d_{\text{trod}} = 20 \cdot \text{mm}$

Root area, $S_{\text{trod}} = 220 \cdot \text{mm}^2$

Number of bolts, $N_{\text{trod}} = 8$

Bolting area provided, $A_b := N_{\text{trod}} \cdot S_{\text{trod}} \quad A_b = 1760 \cdot \text{mm}^2$

To have adequate number and size, A_b shall be greater than A_m

$$A_m = 821.8 \cdot \text{mm}^2 \quad (1=\text{true}, 0=\text{false}) \quad A_b > A_m = 1$$

Bolt load for gasket conditions:

$$W_g := \frac{(A_m + A_b) S_{bg}}{2} \quad W_g = 222039 \cdot \text{N}$$

C- FLANGE DESIGN PROCEDURE. (§ 4.16.7)

C.1 Moment arms for flange loads under operating conditions, in acc. with Table 4.16.6

Radial distance from bolt-circle to point of action of Hydrostatic end force:

$$h_D := \frac{C - B - g_1}{2} \quad h_D = 33.9 \cdot \text{mm}$$

$$h_G := \frac{C - G}{2} \quad h_G = 21.4 \cdot \text{mm}$$

Radial distance from bolt-circle to point of action of Hydrostatic differences:

$$h_T := \frac{1}{2} \cdot \left(\frac{C - B}{2} + h_G \right) \quad h_T = 32.3 \cdot \text{mm}$$

C.2 Forces

Total hydrostatic end force for internal pressure:

$$H := 0.785 G^2 \cdot P \quad H = 7367.7291 \cdot \text{lbf} \quad H = 32773 \cdot \text{N}$$

Hydrostatic end force on area inside of flange for internal pressure:

$$H_D := 0.785 \cdot B^2 \cdot P \quad H_D = 4500.0019 \cdot \text{lbf} \quad H_D = 20017 \cdot \text{N}$$

$$\text{Hydrostatic difference:} \quad H_T := H - H_D \quad H_T = 2867.7272 \cdot \text{lbf} \quad H_T = 12756 \cdot \text{N}$$

$$\text{Bolt Load at Operating Conditions:} \quad H_G := W_o - H \quad H_G = 5182.3437 \cdot \text{lbf} \quad H_G = 23052 \cdot \text{N}$$

Total hydrostatic end force for external pressure:

$$H_{\text{ext}} := 0.785 G^2 \cdot P_{\text{ext}} \quad H_{\text{ext}} = 715.9208 \cdot \text{lbf} \quad H_{\text{ext}} = 3185 \cdot \text{N}$$

Hydrostatic end force on area inside of flange for external pressure:

$$H_{\text{Dext}} := 0.785 \cdot B^2 \cdot P_{\text{ext}} \quad H_{\text{Dext}} = 437.2643 \cdot \text{lbf} \quad H_{\text{Dext}} = 1945 \cdot \text{N}$$

$$\text{Hydrostatic difference:} \quad H_{\text{Ttext}} := H_{\text{ext}} - H_{\text{Dext}} \quad H_{\text{Ttext}} = 278.6565 \cdot \text{lbf} \quad H_{\text{Ttext}} = 1240 \cdot \text{N}$$

$$\text{Bolt Load at Operating Conditions:} \quad H_{\text{Gext}} := W_{\text{oext}} - H_{\text{ext}} \quad H_{\text{Gext}} = 503.5674 \cdot \text{lbf} \quad H_{\text{Gext}} = 2240 \cdot \text{N}$$

C.3 Calculation of Flange Stress Factors, Table 4.16.4 and 4.16.5.

$$K := \frac{A}{B} \quad K = 1.8053$$

$$Y := \frac{1}{K-1} \cdot \left(0.66845 + 5.71690 \cdot \frac{K^2 \cdot \log(K)}{K^2 - 1} \right) \quad Y = 3.4576$$

$$T := \frac{K^2 \cdot (1 + 8.55246 \cdot \log(K)) - 1}{(1.04720 + 1.9448 \cdot K^2) \cdot (K - 1)} \quad T = 1.5822$$

$$U := \frac{K^2 \cdot (1 + 8.55246 \cdot \log(K)) - 1}{1.36136 \cdot (K^2 - 1) \cdot (K - 1)} \quad U = 3.7995$$

$$Z := \frac{K^2 + 1}{K^2 - 1} \quad Z = 1.8853$$

$$h_o := \sqrt{B \cdot g_0} \quad h_o = 32.0 \cdot \text{mm}$$

Numerical decrease of hub length "h" for the F factor calculation in conservative way:

$$h := \begin{cases} h_r & \text{if } \frac{h_r}{h_o} \leq 2 \\ 2 \cdot h_o & \text{otherwise} \end{cases} \quad \begin{aligned} h_r &= 52.8 \cdot \text{mm} \\ h &= 52.8 \cdot \text{mm} \end{aligned}$$

$$X_h := \frac{h}{h_o} \quad X_h = 1.6503$$

$$X_g := \frac{g_1}{g_0} \quad X_g = 2.7955$$

$$\begin{aligned} F := & 0.897697 - 0.297012 \cdot \ln(X_g) + 9.5257 \cdot 10^{-3} \cdot \ln(X_h) \dots \\ & + 0.123586 \cdot (\ln(X_g))^2 + 0.0358580 \cdot (\ln(X_h))^2 - 0.194422 \cdot \ln(X_g) \cdot \ln(X_h) \dots \\ & + -0.0181259 \cdot (\ln(X_g))^3 + 0.0129360 \cdot (\ln(X_h))^3 \dots \\ & + -0.0377693 \cdot (\ln(X_g)) \cdot (\ln(X_h))^2 + 0.0273791 \cdot (\ln(X_g))^2 \cdot \ln(X_h) \end{aligned} \quad F = 0.6233$$

$$V := \begin{cases} \left(0.500244 + \frac{0.227914}{X_g} - 1.87071 \cdot X_h - \frac{0.344410}{X_g^2} + 2.49189 \cdot X_h^2 \dots \right. \\ \left. + 0.873446 \cdot \frac{X_h}{X_g} + \frac{0.189953}{X_g^3} - 1.06082 \cdot X_h^3 - 1.49970 \cdot \frac{X_h^2}{X_g} \dots \right. \\ \left. + 0.719413 \cdot \frac{X_h}{X_g^2} \right) & \text{if } 0.1 \leq X_h \leq 0.5 \\ \left(0.0144868 - \frac{0.135977}{X_g} - \frac{0.0461919}{X_h} + \frac{0.560718}{X_g^2} + \frac{0.0529829}{X_h^2} \dots \right. \\ \left. + \frac{0.244313}{X_g \cdot X_h} + \frac{0.113929}{X_g^3} - \frac{0.00928265}{X_h^3} - \frac{0.0266293}{X_g \cdot X_h^2} - \frac{0.217008}{X_g^2 \cdot X_h} \right) & \text{if } 0.5 < X_h \leq 2 \\ \text{"ERROR"} & \text{otherwise} \end{cases}$$

$V = 0.0648$

$$f := \max \left[1, \left[\frac{\left[0.0927779 - 0.0336633 \cdot X_g + 0.964176 \cdot X_g^2 \dots \right. \right.}{\left[1 - 5.96093 \cdot 10^{-3} \cdot X_g + 1.62904 \cdot X_h \dots \right.} \right. \right]$$

$f = 1.0000$

$$d := \frac{U \cdot g_0^2 \cdot h_o}{V}$$

$d = 81661 \cdot \text{mm}^3$

$$e := \frac{F}{h_o}$$

$e = 19.4816 \frac{1}{\text{m}}$

$$L := \frac{t \cdot e + 1}{T} + \frac{t^3}{d}$$

$L = 1.0935$

C.4 Calculation of the Flange Moments of Inertia Table 4.16.7

$$I := \frac{0.0874 \cdot L \cdot g_0^2 \cdot h_o \cdot B}{V}$$

$$I = 318589.9 \cdot \text{mm}^4$$

$$G_{\text{avg}} := 0.5 \cdot (g_0 + g_1)$$

$$G_{\text{avg}} = 12.5 \cdot \text{mm}$$

$$A_R := 0.5 \cdot (A - B)$$

$$A_R = 62.5 \cdot \text{mm}$$

$$A_A := \begin{cases} A_R & \text{if } t \geq G_{\text{avg}} \\ (h + t) & \text{if } t < G_{\text{avg}} \end{cases}$$

$$A_A = 62.5 \cdot \text{mm}$$

$$B_B := \begin{cases} t & \text{if } t \geq G_{\text{avg}} \\ G_{\text{avg}} & \text{if } t < G_{\text{avg}} \end{cases} = 0.0239$$

$$B_B = 23.9 \cdot \text{mm}$$

$$C_C := \begin{cases} h & \text{if } t \geq G_{\text{avg}} \\ A_R - G_{\text{avg}} & \text{if } t < G_{\text{avg}} \end{cases}$$

$$C_C = 52.8 \cdot \text{mm}$$

$$D_{\text{DG}} := \begin{cases} G_{\text{avg}} & \text{if } t \geq G_{\text{avg}} \\ t & \text{if } t < G_{\text{avg}} \end{cases}$$

$$D_{\text{DG}} = 12.5 \cdot \text{mm}$$

$$K_{AB} := A_A \cdot B_B^3 \cdot \left[\frac{1}{3} - 0.21 \cdot \frac{B_B}{A_A} \cdot \left[1 - \frac{1}{12} \cdot \left(\frac{B_B}{A_A} \right)^4 \right] \right]$$

$$K_{CD} := C_C \cdot D_{\text{DG}}^3 \cdot \left[\frac{1}{3} - 0.105 \cdot \frac{D_{\text{DG}}}{C_C} \cdot \left[1 - \frac{1}{192} \cdot \left(\frac{D_{\text{DG}}}{C_C} \right)^4 \right] \right]$$

$$I_P := K_{AB} + K_{CD}$$

$$I_P = 247788.6 \cdot \text{mm}^4$$

C.5 Moments

External moment

$$M_{oe} := 4 \cdot M_E \cdot \left(\frac{I}{0.3846 \cdot I_P + I} \right) \cdot \frac{h_D}{C - 2 \cdot h_D} + F_A \cdot h_D$$

$$M_{oe} = 5874.5 \cdot \text{lbf} \cdot \text{in}$$

$$M_{oe} = 663726 \cdot \text{N} \cdot \text{mm}$$

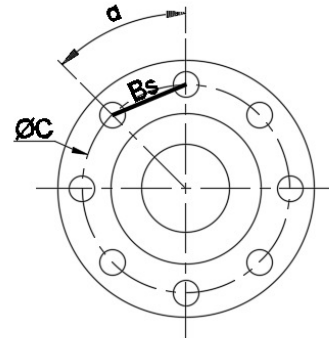
Bolt Spacing Equations Table 4.16.11

Bolt circle diameter : $C = 9.5 \cdot \text{in}$

$$C = 241.3 \cdot \text{mm}$$

Angle between 2 bolts : $\alpha := \frac{2 \cdot \pi}{N_{\text{trod}}}$

$$\alpha = 45 \cdot \text{deg}$$



Bolt spacing : $B_s := C \cdot \sin\left(\frac{\alpha}{2}\right)$ $B_s = 3.6355 \cdot \text{in}$

$$B_s = 92.3 \cdot \text{mm}$$

Nota : The bolt spacing used is the chord length between adjacent bolt locations

Maximum bolt spacing according to Appendix 2 §2.5(d) eq.3 of Section VIII Div.1 :

MANDATORY only for lethal.service.

$$B_{s\max} := 2 \cdot d_{\text{trod}} + \frac{6 \cdot t}{m_{gkt} + 0.5} = 3.8331 \cdot \text{in}$$

$$B_{s\max} = 97.4 \cdot \text{mm}$$

$$B_s < B_{s\max} = 1$$

with : $B_{sc} := \max\left(1, \sqrt{\frac{B_s}{2d_{\text{trod}} + t}}\right) = 1.2021$

otherwise Bolt spacing correction factor : $B_{sc} := 1$ **for non lethal service**

Moment acting on the flange due to hydrostatic end force for internal pressure

$$M_{\text{oint}} := F_s \cdot [(H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G) \cdot B_{sc} + M_{oe}]$$

$$M_{\text{oint}} = 2247103 \cdot \text{N} \cdot \text{mm}$$

With $F_s = 1.0$ for non-split rings (non split loose type flanges)

$$M_{\text{oint}} = 19888.5 \cdot \text{in} \cdot \text{lbf}$$

Moment acting on the flange due to bolting up condition for internal pressure

$$M_{\text{gint}} := \frac{W_g \cdot (C - G) \cdot B_{sc} \cdot F_s}{2}$$

$$M_{\text{gint}} = 4756111 \cdot \text{N} \cdot \text{mm}$$

$$M_{\text{gint}} = 42095.1 \cdot \text{in} \cdot \text{lbf}$$

Moment acting on the flange due to hydrostatic end force for external pressure

$$M_{\text{oext}} := |F_s \cdot [H_{\text{Dext}} \cdot (h_D - h_G) + H_{\text{Text}} \cdot (h_T - h_G) + M_{oe}]|$$

$$M_{\text{oext}} = 701388 \cdot \text{N} \cdot \text{mm}$$

$F_s = 1.0$ for non-split rings (non split loose type flanges)

$$M_{\text{oext}} = 6207.8 \cdot \text{in} \cdot \text{lbf}$$

Moment acting on the flange due to bolting up condition for external pressure

$$M_{\text{gext}} := W_g \cdot h_G \cdot F_s$$

$$M_{\text{gext}} = 42095.1 \cdot \text{in} \cdot \text{lbf}$$

$$M_{\text{gext}} = 4756110.7 \cdot \text{N} \cdot \text{mm}$$

Maximal moments :

$$M_o := \max(M_{\text{oext}}, M_{\text{oint}}) = 19888.5 \cdot \text{in} \cdot \text{lbf}$$

$$M_g := \max(M_{\text{gext}}, M_{\text{gint}}) = 42095.1 \cdot \text{in} \cdot \text{lbf}$$

$$M_o = 2.2 \times 10^6 \cdot \text{N} \cdot \text{mm}$$

$$M_g = 4.8 \times 10^6 \cdot \text{N} \cdot \text{mm}$$

C.6 Calculation of Flange stresses in Operating and Bolting Up conditions, Table 4.16.8

Longitudinal Hub Stress in Flange

$$S_{H.o} := \frac{f \cdot M_o}{L \cdot g_l^2 \cdot B} \quad S_{H.o} = 38.9 \cdot \text{MPa}$$

$$S_{H.g} := \frac{f \cdot M_g}{L \cdot g_l^2 \cdot B} \quad S_{H.g} = 82.4 \cdot \text{MPa}$$

Radial Stress in Flange

$$S_{R.o} := \frac{(1.33 \cdot t \cdot e + 1) \cdot M_o}{L \cdot t^2 \cdot B} \quad S_{R.o} = 37.6 \cdot \text{MPa}$$

$$S_{R.g} := \frac{(1.33 \cdot t \cdot e + 1) \cdot M_g}{L \cdot t^2 \cdot B} \quad S_{R.g} = 79.5 \cdot \text{MPa}$$

Tangential Stress in Flange

$$S_{T.o} := \frac{Y \cdot M_o}{t^2 \cdot B} - Z \cdot S_{R.o} \quad S_{T.o} = 16.9 \cdot \text{MPa}$$

$$S_{T.g} := \frac{Y \cdot M_g}{t^2 \cdot B} - Z \cdot S_{R.g} \quad S_{T.g} = 35.7 \cdot \text{MPa}$$

C.7 Allowable flange design stresses at Operating and Bolting Up condition

Criteria evaluation
(1=true, 0=false)

	Stress values	Allowable Stresses	
Longitudinal Hub Stress	$S_{H.o} = 38.9 \cdot \text{MPa}$	$\min(1.5 \cdot S_{fo}, 2.5 \cdot S_{no}) = 138.4 \cdot \text{MPa}$	$S_{H.o} \leq \min(1.5 \cdot S_{fo}, 2.5 \cdot S_{no}) = 1$
	$S_{H.g} = 82.4 \cdot \text{MPa}$	$\min(1.5 \cdot S_{fg}, 2.5 \cdot S_{ng}) = 172.5 \cdot \text{MPa}$	$S_{H.g} \leq \min(1.5 \cdot S_{fg}, 2.5 \cdot S_{ng}) = 1$
Radial Hub Stress	$S_{R.o} = 37.6 \cdot \text{MPa}$	$S_{fo} = 92.3 \cdot \text{MPa}$	$S_{R.o} \leq S_{fo} = 1$
	$S_{R.g} = 79.5 \cdot \text{MPa}$	$S_{fg} = 115 \cdot \text{MPa}$	$S_{R.g} \leq S_{fg} = 1$
Tangential Hub Stress	$S_{T.o} = 16.9 \cdot \text{MPa}$	$S_{fo} = 92.3 \cdot \text{MPa}$	$S_{T.o} \leq S_{fo} = 1$
	$S_{T.g} = 35.7 \cdot \text{MPa}$	$S_{fg} = 115 \cdot \text{MPa}$	$S_{T.g} \leq S_{fg} = 1$
Combined Hub Stress 1	$\frac{S_{H.o} + S_{R.o}}{2} = 38.2 \cdot \text{MPa}$	$S_{fo} = 92.3 \cdot \text{MPa}$	$\frac{S_{H.o} + S_{R.o}}{2} \leq S_{fo} = 1$
	$\frac{S_{H.g} + S_{R.g}}{2} = 80.9 \cdot \text{MPa}$	$S_{fg} = 115 \cdot \text{MPa}$	$\frac{S_{H.g} + S_{R.g}}{2} \leq S_{fg} = 1$

Combined Hub Stress 2	$\frac{S_{H.o} + S_{T.o}}{2} = 27.9 \cdot \text{MPa}$	$S_{fo} = 92.3 \cdot \text{MPa}$	$\frac{S_{H.o} + S_{T.o}}{2} \leq S_{fo} = 1$
	$\frac{S_{H.g} + S_{T.g}}{2} = 59.1 \cdot \text{MPa}$	$S_{fg} = 115 \cdot \text{MPa}$	$\frac{S_{H.g} + S_{T.g}}{2} \leq S_{fg} = 1$

C.8 Flange rigidity criterion

Modulus of Elasticity at operating temperature:

$$E_{yo} = 28800000 \cdot \text{MPa}$$

Modulus of Elasticity at gasket seating condition

$$E_{yg} = 29400000 \cdot \text{MPa}$$

Rigidity index factor:

$$K_R := 0.3$$

$$J_o := \frac{52.14 \cdot V \cdot M_o}{L \cdot E_{yo} \cdot g_0^2 \cdot K_R \cdot h_o}$$

$$J_o = 0.0006$$

$$J_g := \frac{52.14 \cdot V \cdot M_g}{L \cdot E_{yg} \cdot g_0^2 \cdot K_R \cdot h_o}$$

$$J_g = 0.0012$$

(1=true, 0=false)

J_o shall be smaller or equal than 1.0

$$J_o \leq 1.0 = 1$$

J_g shall be smaller or equal than 1.0

$$J_g \leq 1.0 = 1$$



APPENDIX 1

(IMPACT TEST REQUIREMENTS UG 20 (f))

Minimum design metal temperature : $T_{\min} = 0^{\circ}\text{C}$

PARTS	METAL	IMPACT TEST
FLAT HEADS nominal thick : 70 mm	SA 516 gr 60 Normalized Governing Thick : 17.5 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
LONGITUDINAL GIRDER nominal thick : 70 mm type : CP40	SA 516 gr 60 Normalized Governing Thick : 17.5 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
PANEL Aa nominal thick : 80 mm	SA 516 gr 60 Normalized Governing Thick : 20 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
PANEL Ab nominal thick : 70 mm	SA 516 gr 60 Normalized Governing Thick : 17.5 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
PANEL Ba nominal thick : 80 mm	SA 516 gr 60 Normalized Governing Thick : 20 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D
PANEL Bb nominal thick : 70 mm	SA 516 gr 60 Normalized Governing Thick : 17.5 mm (1/4 nominal thickness)	EXEMPTED ACCORDING TO FIG. UCS 66M CURVE D

Plate governing thick =

plate without nozzles : $\frac{1}{4}$ nominal thickness (UCS 66 (a)(3))

plate with nozzles : the larger of $\frac{1}{4}$ panel nominal thickness and nozzle pipe
thickness (UCS 66 (a)(1)(c))

THREAD ROD [Dia <= 64 mm]	SA 193 gr B7	IMPACT TEST EXEMPTION ACCORDING TO FIG. UCS 66M NOTE (c) ABOVE -55°F (-48°C)
NUTS	SA 194 gr 7	IMPACT TEST EXEMPTION ACCORDING TO FIG. UCS 66M NOTE (c) ABOVE -55°F (-48°C)
NON CODE EXCHANGE PLATE	SA 240 type 316L Governing Thick : 1 mm	EXEMPTED PER UHA 51
WELDS	Weld metal + Heat affected zone (Note : Mandatory in all cases if MDMT < - 29°C)	NOT REQUIRED ACCORDING TO UCS 67 (a) WHEN IMPACT TEST IS NOT REQUIRED ON BASE METAL
	Impact test on production test plate	NOT REQUIRED PER UG 84
NOZZLE FLANGE ASME B16-5	SA 182 F 316L	NOT REQUIRED ACC. TO UHA 51
NOZZLE PIPE NPS 6"	SA 312 TP 316L Nominal thick : 7.11 mm (sch.40 s)	EXEMPTED PER UHA 51
LININGS	SA 240 type 316L Thickness = 3 mm	EXEMPTED PER UHA 51



APPENDIX 2

TEST PRESSURE CALCULATION ACCORDING TO ASME VIII Div.1 CODE UG-99 (b)

Test pressure = 1.3 x Design pressure x Lowest ratio (S at test temperature / S at design temperature)

According to the configuration of hydrostatic test carry out on exchanger (Manometer on the top of the CP), we use for the definition of test pressure the design pressure without the pressure static head of liquid.

Design pressure Side A :	$P_{designA} = 1.04 \cdot \text{MPa}$	Design pressure Side B :	$P_{designB} = 1.04 \cdot \text{MPa}$
	$P_{designA} = 10.4 \cdot \text{bar}$		$P_{designB} = 10.4 \cdot \text{bar}$

<u>Bolting Side A (Tie rod)</u>	:	Mat _{trod} = "SA 193 gr B7"	<u>Ratio</u>
Allowable stress at test temperature :		Sa _{trod} = 172·MPa	
Allowable stress at design temperature :		Sb _{trod} = 172·MPa	$\frac{Sa_{trod}}{Sb_{trod}} = 1$

<u>Bolting Side B (Tie rod)</u>	:	Mat _{trod} = "SA 193 gr B7"	<u>Ratio</u>
Allowable stress at test temperature :		Sa _{trod.B} = 172·MPa	
Allowable stress at design temperature :		Sb _{trod.B} = 172·MPa	$\frac{Sa_{trod.B}}{Sb_{trod.B}} = 1$

Bolting ratio is not included in the test pressure calculation.

<u>Panels side A</u>	:	Mat _{panel} = "SA 516 gr 60N"	
Allowable stress at test temperature :		Sa _{panel} = 118·MPa	
Allowable stress at design temperature :		Sb _{panel} = 118·MPa	$\frac{Sa_{panel}}{Sb_{panel}} = 1$

<u>Panels side B</u>	:	Mat _{panel} = "SA 516 gr 60N"	
Allowable stress at test temperature :		Sa _{panel.B} = 118·MPa	
Allowable stress at design temperature :		Sb _{panel.B} = 118·MPa	$\frac{Sa_{panel.B}}{Sb_{panel.B}} = 1$

<u>Head</u>	:	Mat _{head} = "SA 516 gr 60N"	
Allowable stress at test temperature :		Sa _{head} = 118·MPa	
Allowable stress at design temperature :		Sb _{head} = 118·MPa	$\frac{Sa_{head}}{Sb_{head}} = 1$

<u>Columns</u>	:	Mat _{columns} = "SA 516 gr 60N"	
Allowable stress at test temperature :		Sa _{col} = 118·MPa	
Allowable stress at design temperature :		Sb _{col} = 118·MPa	$\frac{Sa_{col}}{Sb_{col}} = 1$

<u>Pipe A1</u>	:	Mat _{PipeA1} = "SA 312 TP 316L"	
Allowable stress at test temperature :		Sa _{PipeA1} = 115·MPa	$\frac{Sa_{PipeA1}}{Sb_{PipeA1}} = 1$
Allowable stress at design temperature	:	Sb _{PipeA1} = 115·MPa	
<u>Flange A1</u>	:	Mat _{FlangeA1} = "SA 182 F 316L"	
Allowable stress at test temperature :		Sa _{FlangeA1} = 115·MPa	$\frac{Sa_{FlangeA1}}{Sb_{FlangeA1}} = 1.2459$
Allowable stress at design temperature	:	Sb _{FlangeA1} = 92.3·MPa	
<u>Pipe A2</u>	:	Mat _{PipeA2} = "SA 312 TP 316L"	
Allowable stress at test temperature :		Sa _{PipeA2} = 115·MPa	$\frac{Sa_{PipeA2}}{Sb_{PipeA2}} = 1$
Allowable stress at design temperature	:	Sb _{PipeA2} = 115·MPa	
<u>Flange A2</u>	:	Mat _{FlangeA2} = "SA 182 F 316L"	
Allowable stress at test temperature :		Sa _{FlangeA2} = 115·MPa	$\frac{Sa_{FlangeA2}}{Sb_{FlangeA2}} = 1.2459$
Allowable stress at design temperature	:	Sb _{FlangeA2} = 92.3·MPa	

<u>Pipe B1</u>	:	Mat _{PipeB1} = "SA 312 TP 316L"	$\frac{S_{aPipeB1}}{S_{bPipeB1}} = 1$
Allowable stress at test temperature :		S _{aPipeB1} = 115·MPa	
Allowable stress at design temperature	:	S _{bPipeB1} = 115·MPa	
<u>Flange B1</u>	:	Mat _{FlangeB1} = "SA 182 F 316L"	$\frac{S_{aFlangeB1}}{S_{bFlangeB1}} = 1.2459$
Allowable stress at test temperature :		S _{aFlangeB1} = 115·MPa	
Allowable stress at design temperature	:	S _{bFlangeB1} = 92.3·MPa	
<u>Pipe B2</u>	:	Mat _{PipeB2} = "SA 312 TP 316L"	$\frac{S_{aPipeB2}}{S_{bPipeB2}} = 1$
Allowable stress at test temperature :		S _{aPipeB2} = 115·MPa	
Allowable stress at design temperature	:	S _{bPipeB2} = 115·MPa	
<u>Flange B2</u>	:	Mat _{FlangeB2} = "SA 182 F 316L"	$\frac{S_{aFlangeB2}}{S_{bFlangeB2}} = 1.2459$
Allowable stress at test temperature :		S _{aFlangeB2} = 115·MPa	
Allowable stress at design temperature	:	S _{bFlangeB2} = 92.3·MPa	

Plates or Linings : $Mat_{plates} = \text{"SA 240 type 316L"}$
 Allowable stress at test temperature : $Sa_{plates} = 115 \cdot \text{MPa}$
 Allowable stress at design temperature : $Sb_{plates} = 92.3 \cdot \text{MPa}$

$$\frac{Sa_{plates}}{Sb_{plates}} = 1.2459$$

Linings side A : $Mat_{linings.A} = \text{"SA 240 type 316L"}$
 Allowable stress at test temperature : $Sa_{linings.A} = 115 \cdot \text{MPa}$
 Allowable stress at design temperature : $Sb_{linings.A} = 92.3 \cdot \text{MPa}$

$$\frac{Sa_{linings.A}}{Sb_{linings.A}} = 1.2459$$

Linings side B : $Mat_{linings.B} = \text{"SA 240 type 316L"}$
 Allowable stress at test temperature : $Sa_{linings.B} = 115 \cdot \text{MPa}$
 Allowable stress at design temperature : $Sb_{linings.B} = 92.3 \cdot \text{MPa}$

$$\frac{Sa_{linings.B}}{Sb_{linings.B}} = 1.2459$$

$$Lowest_{ratio.A} = 1$$

$$Lowest_{ratio.B} = 1$$

Nota : If Test pressure calculated below is greater than $1.3 \cdot Lowest_{ratio} \cdot P_{design}$ check the calculation done with 90% of yield constraints according to ASME VIII div.1 CODE UG-99(c)

$$Coef_{maxA} = 1.3$$

$$Coef_{maxB} = 1.3$$

Adopted Test Pressure

$$Test_{pressure.A} = Coef_{maxA} \cdot P_{designA}$$

$$Test_{pressure.B} = Coef_{maxB} \cdot P_{designB}$$

$$Test_{pressure.A} = 1.36 \cdot \text{MPa}$$

$$Test_{pressure.B} = 1.36 \cdot \text{MPa}$$

$$Test_{pressure.A} = 13.6 \cdot \text{bar}$$

$$Test_{pressure.B} = 13.6 \cdot \text{bar}$$

$$Test_{pressure.A} = 197 \cdot \text{psi}$$

$$Test_{pressure.B} = 197 \cdot \text{psi}$$



APPENDIX 3

FULL VACUUM CALCULATION NOTE

1. SQUARE HEADS

Since these parts are unstayed flat heads designed previously at $P = 1.06 \cdot \text{MPa}$ (for internal pressure), no new calculation is necessary for full vacuum (15 psi, 1.03 bar, 0.103MPa).

2. STEEL PANELS (applicable if analytical calculation done)

Since these parts are unstayed flat heads designed previously at $P = 1.06 \cdot \text{MPa}$ (for internal pressure on side A and on side B), no new calculation is necessary for full vacuum (15 psi, 1.03 bar 0.103MPa).

3. NOZZLES PIPES LINING

Calculations according to ASME VIII Edition 2019, UG 28

3.1.1 - Pipe size :

Item : Item_{opening} = "A1-A2-B1-B2"

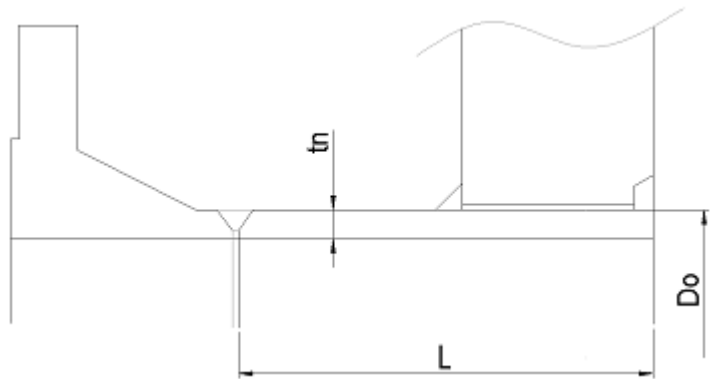
Pipe material : Mat_{pipe} = "SA 312 TP 316L"

Nominal diameter : N_{Dia} = "NPS 6"

External diameter : D_o = 168.3·mm
(in corroded condition)

Length of pipe : L = 165·mm

Thickness of pipe : t_n = 6.6·mm
(in corroded condition)



Step 1 - Ratio determination :

$$\frac{L}{D_o} = 0.9804 \quad \frac{D_o}{t_n} = 25.5$$

Step 2,3 - Factor A determination according to ASME II Part D - Subpart 3, Fig.G

$$A = 0.011134$$

Step 4,5 - Factor B determination according to ASME II Part D - Subpart 3,

Fig = "HA-4"

$$B = 82 \cdot \text{MPa}$$

Step 6 - Calculation of Maximun allowable external pressure.

- For pipe having value of $\frac{D_o}{t_n} \geq 10$

Value of P_a

$$P_a := 4 \cdot \frac{B}{3 \left(\frac{D_o}{t_n} \right)}$$

$$P_a = 4.3 \cdot \text{MPa}$$

- For pipe having value of $\frac{D_o}{t_n} < 10$

Value of P_{a1}

$$P_{a1} := \left(\frac{2.167}{\frac{D_o}{t_n}} - 0.0833 \right) \cdot B$$

$$P_{a1} = 0.1 \cdot \text{MPa}$$

(P_{a1} value can not be lower than 0 MPa)

Value of P_{a2}

Allowable stress (Pipe)

$$S_m = 115 \cdot \text{MPa} \quad (\text{ASME II part D - Table 1-A})$$

Yield strength value (Pipe)

$$S_y = 139 \cdot \text{MPa} \quad (\text{ASME II part D - Table Y-1})$$

$$S := \min(2 \cdot S_m, 0.9 \cdot S_y)$$

$$S = 125.1 \cdot \text{MPa}$$

$$P_{a2} := \frac{2 \cdot S}{\frac{D_o}{t_n}} \cdot \left(1 - \frac{1}{\frac{D_o}{t_n}} \right)$$

$$P_{a2} = 9.4 \cdot \text{MPa}$$

- Result of the calculation:

$$P_{\text{ext}} := \begin{cases} P_a & \text{if } \frac{D_o}{t_n} \geq 10 \\ \min(P_{a1}, P_{a2}) & \text{otherwise} \end{cases}$$

$$P_{\text{ext}} = 4.3 \cdot \text{MPa}$$

External pressure P_{ext} shall be upper or equal than required external working pressure $P_{\text{UG28.f}}$

$$P_{\text{UG28.f}} = 0.103 \cdot \text{MPa}$$

(1=true, 0=false)

$$P_{\text{ext}} \geq P_{\text{UG28.f}} = 1$$

$$P_{\text{ext}} = 4.3 \cdot \text{MPa}$$

4. PANEL LINING SIDE A

Calculation of panels linings in full vacuum condition according to ASME VIII Div.1, Appendix 17.

Lining material : $Mat_{\text{lining}} = \text{"SA 240 type 316L"}$

Lining thickness : $Th_{\text{lining}} = 3 \cdot \text{mm}$

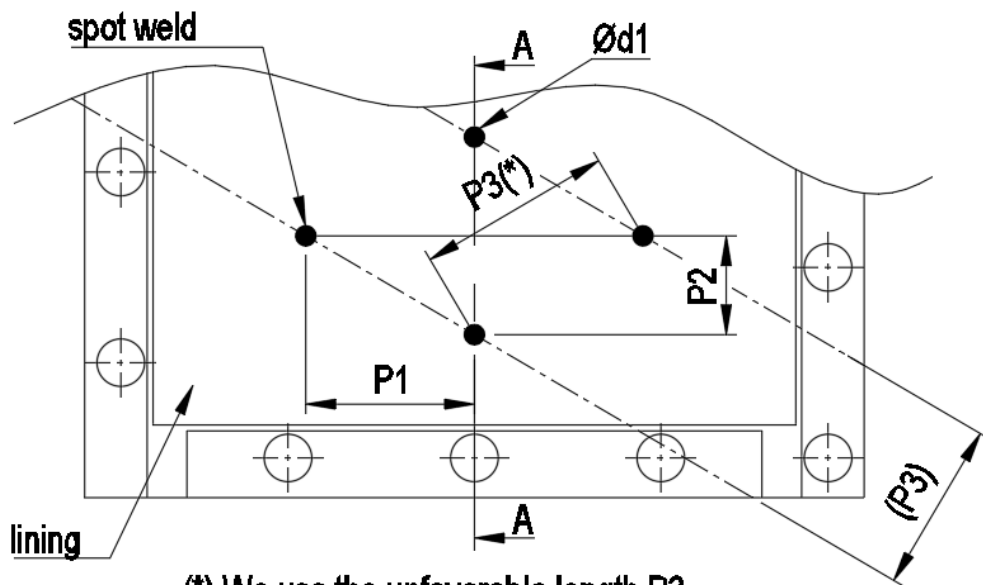
Maximum allowable stress at design Temperature : $S_{\text{lining}} = 92.3 \cdot \text{MPa}$

Thickness limitations (Appendix 17, App. 17-4) : $Th_{\text{min}} := 0.8 \text{mm}$ (1=true, 0=false)

The nominal thickness Th_{lining} shall be not less than Th_{min} $Th_{\text{lining}} \geq Th_{\text{min}} = 1$

Lining attachment :

Lining composed of one sheet attached to side panels by spot welds symmetrically spaced on a rectangular pitch (Appendix 17, Fig. 17-4)



(*) We use the unfavorable length P3 instead of (P3) to be more conservative

$$p_1 = 85 \cdot \text{mm} \quad p_2 = 50 \cdot \text{mm} \quad p_3 = 98.6 \cdot \text{mm} \quad p := \max(p_1, p_2, p_3) \quad p = 98.6 \cdot \text{mm}$$

Maximum allowable working pressure App. 17-5 (b)(2)

Calculation of maximum allowable working pressure P according to UG-47 equation (2) :

$$P_{fv} := \frac{Th_{\text{lining}}^2 \cdot S_{\text{lining}} \cdot C_{\text{lining}}}{p^2} \quad P_{fv} = 0.179 \cdot \text{MPa}$$

with : $C_{\text{lining}} = 2.1$

$p = 98.6 \cdot \text{mm}$ maximum pitch = greater of p_1, p_2, p_3

$Th_{\text{lining}} = 3 \cdot \text{mm}$

$S_{\text{lining}} = 92.3 \cdot \text{MPa}$

External pressure P_{fv} shall be upper than required external working pressure $P_{UG28.f}$

$P_{UG28.f} = 0.103 \cdot \text{MPa}$

(1=true, 0=false)

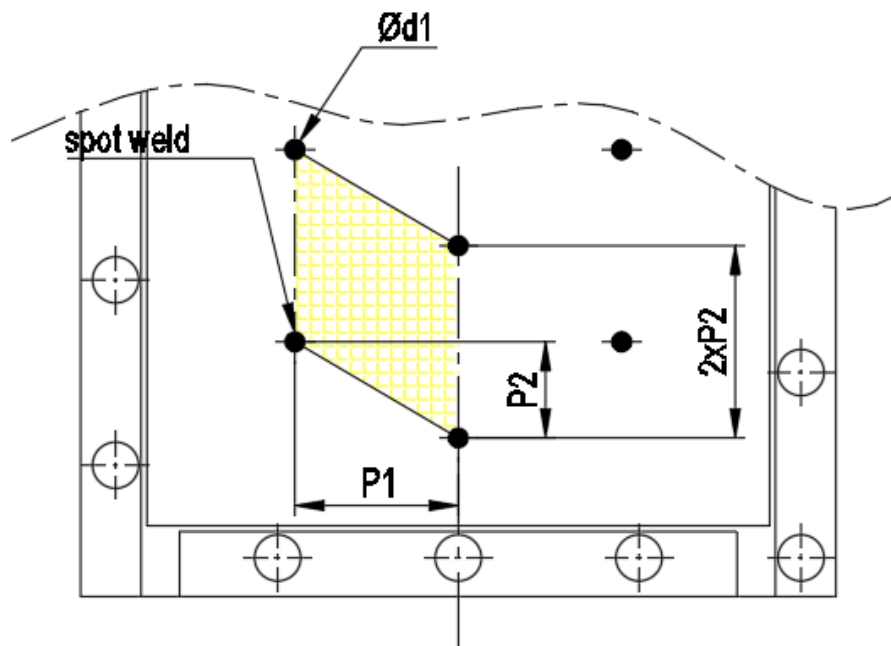
$P_{fv} \geq P_{UG28.f} = 1$

$P_{fv} = 0.179 \cdot \text{MPa}$

Spot welds verification :

$$p_1 = 85 \cdot \text{mm}$$

$$p_2 = 50 \cdot \text{mm}$$



Considered an area : $S_{fv} := 2 \cdot p_2 \cdot p_1$ $S_{fv} = 8500 \cdot \text{mm}^2$

Design pressure : $p_{fv} := P_{UG28.f}$ $p_{fv} = 0.103 \cdot \text{MPa}$

Load due to pressure on this area : $F_{fv} := S_{fv} \cdot p_{fv}$ $F_{fv} = 875.5 \cdot \text{N}$

Strength section of spot/screw welds supporting this area :

1 spot weld with a diameter of $d_1 = 6 \cdot \text{mm}$

$$d_1 = 6 \cdot \text{mm}$$

Allowable stress at design Temperature $S_{\text{lining}} = 92.3 \cdot \text{MPa}$

$$S_{\text{lining}} = 92.3 \cdot \text{MPa}$$

Tension area of spot weld $S_T := \frac{d_1^2 \cdot \pi}{4}$

$$S_T = 28.3 \cdot \text{mm}^2$$

Shear area in lining around the screw head / spot weld

$$S_s := \pi \cdot d_1 \cdot Th_{\text{lining}} = 56.5 \cdot \text{mm}^2$$

$$S_s = 56.5 \cdot \text{mm}^2$$

Remark: in conservative way, in both case, the minimal diameter d1 is chosen for the shear calculation.

tension stress in spot/screw weld section : $\sigma := \frac{S_{fv} \cdot p_{fv}}{S_T} = 31 \cdot \text{MPa}$

$$\sigma = 31 \cdot \text{MPa}$$

$$\sigma < S_{\text{lining}} = 1$$

Shear stress on lining :

$$\tau := \frac{S_{fv} \cdot p_{fv}}{S_s} = 15.5 \cdot \text{MPa}$$

$$\tau = 15.5 \cdot \text{MPa}$$

$$\tau < 0.8 S_{\text{lining}} = 1$$

Screw verification :

Load by screw : $F_{fv} = 875.5 \cdot N$

Filet shear surface in 1 screw : $S_{f_{\text{filet}}} := \pi \cdot d_{\text{filet}} \cdot \frac{L_{\text{filet}}}{2}$ $S_{f_{\text{filet}}} = 106 \cdot \text{mm}^2$

with : Diameter of thread pitch $d_{\text{filet}} = 9 \cdot \text{mm}$

Length screwed in the panel $L_{\text{filet}} = 7.5 \cdot \text{mm}$

Actual shear stress in filets : $Sh_{\text{filet}} := \frac{F_{fv}}{S_{f_{\text{filet}}}}$ $Sh_{\text{filet}} = 8.3 \cdot \text{MPa}$

Sh_{filet} value shall be smaller than $0.8 \times S_{\text{screw}}$ and $0.8 \times S_{\text{panel}}$ according to UG-23(f)

Allowable stress value of panel : $S_{\text{panel}} = 118 \cdot \text{MPa}$ $Sh_{\text{filet}} < 0.8 \cdot S_{\text{panel}} = 1$

(1=true, 0=false)

Allowable stress value of screw : $S_{\text{screw}} = 92.3 \cdot \text{MPa}$ $Sh_{\text{filet}} < 0.8 \cdot S_{\text{screw}} = 1$

5. PANEL LINING SIDE B

Calculation of panels linings in full vacuum condition according to ASME VIII Div.1, Appendix 17.

Lining material : $Mat_{\text{lining}} = \text{"SA 240 type 316L"}$

Lining thickness : $Th_{\text{lining}} = 3 \cdot \text{mm}$

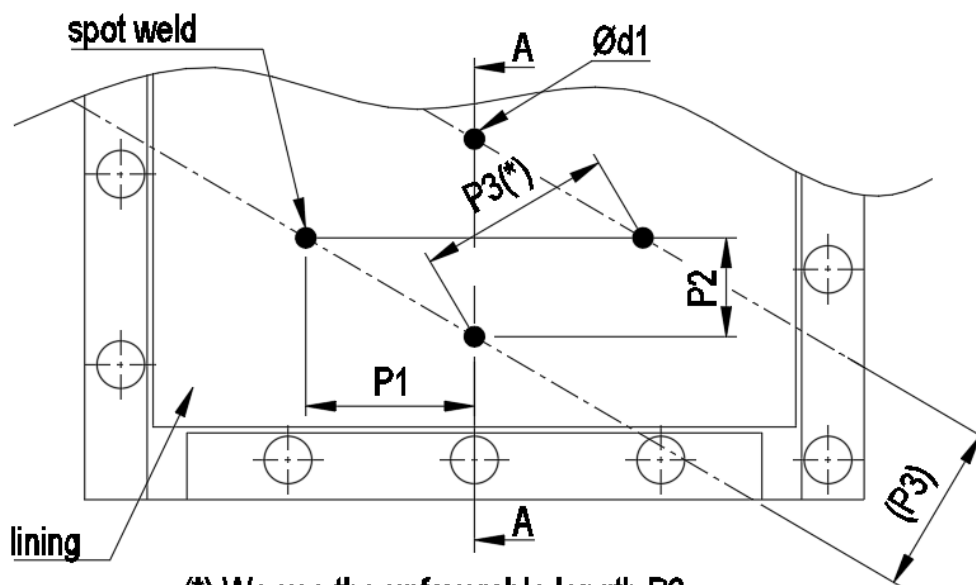
Maximum allowable stress at design Temperature : $S_{\text{lining}} = 92.3 \cdot \text{MPa}$

Thickness limitations (Appendix 17, App. 17-4) : $Th_{\text{min}} := 0.8 \text{mm}$ (1=true, 0=false)

The nominal thickness Th_{lining} shall be not less than Th_{min} $Th_{\text{lining}} \geq Th_{\text{min}} = 1$

Lining attachment :

Lining composed of one sheet attached to side panels by spot welds symmetrically spaced on a rectangular pitch (Appendix 17, Fig. 17-4)



(*) We use the unfavorable length P3 instead of (P3) to be more conservative

$$p_1 = 85 \cdot \text{mm} \quad p_2 = 50 \cdot \text{mm} \quad p_3 = 98.6 \cdot \text{mm} \quad p := \max(p_1, p_2, p_3) \quad p = 98.6 \cdot \text{mm}$$

Maximum allowable working pressure App. 17-5 (b)(2)

Calculation of maximum allowable working pressure P according to UG-47 equation (2) :

$$P_{fv} := \frac{Th_{\text{lining}}^2 \cdot S_{\text{lining}} \cdot C_{\text{lining}}}{p^2} \quad P_{fv} = 0.179 \cdot \text{MPa}$$

with : $C_{\text{lining}} = 2.1$

$p = 98.6 \cdot \text{mm}$ maximum pitch = greater of $p_1, 2p_2, p_3$

$Th_{\text{lining}} = 3 \cdot \text{mm}$

$S_{\text{lining}} = 92.3 \cdot \text{MPa}$

External pressure P_{fv} shall be upper than required external working pressure $P_{UG28.f}$

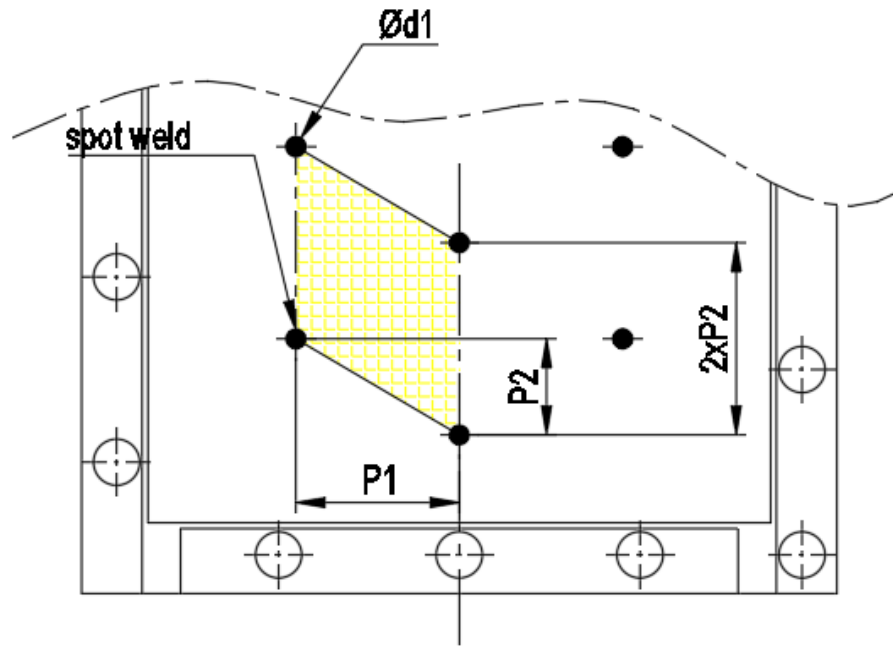
$$P_{UG28.f} = 0.103 \cdot \text{MPa} \quad (1=true, 0=false) \quad P_{fv} \geq P_{UG28.f} = 1$$

$$P_{fv} = 0.179 \cdot \text{MPa}$$

Spot welds verification :

$$p_1 = 85 \cdot \text{mm}$$

$$p_2 = 50 \cdot \text{mm}$$



Considered an area : $S_{fv} := 2 \cdot p_2 \cdot p_1$ $S_{fv} = 8500 \cdot \text{mm}^2$

Design pressure : $p_{fv} := P_{UG28.f}$ $p_{fv} = 0.103 \cdot \text{MPa}$

Load due to pressure on this area : $F_{fv} := S_{fv} \cdot p_{fv}$ $F_{fv} = 875.5 \cdot \text{N}$

Strength section of spot/screw welds supporting this area :

1 spot weld with a diameter of : $d_1 = 6 \cdot \text{mm}$

$d_1 = 6 \cdot \text{mm}$

Allowable stress at design Temperature $S_{\text{lining}} = 92.3 \cdot \text{MPa}$

$S_{\text{lining}} = 92.3 \cdot \text{MPa}$

Tension area of spot weld $S_T := \frac{d_1^2 \cdot \pi}{4}$

$S_T = 28.2743 \cdot \text{mm}^2$

Shear area in lining around the screw head / spot weld $S_s := \pi \cdot d_1 \cdot Th_{\text{lining}} = 56.5487 \cdot \text{mm}^2$

$S_s = 56.5487 \cdot \text{mm}^2$

Remark: in conservative way, in both case, the minimal diameter d1 is chosen for the shear calculation.

tension stress in spot/screw weld section : $\sigma := \frac{F_{fv}}{S_T} = 31 \cdot \text{MPa}$

$$\sigma = 31 \cdot \text{MPa}$$

$$\sigma < S_{\text{lining}} = 1$$

Shear stress on lining : $\tau := \frac{F_{fv}}{S_s} = 15.5 \cdot \text{MPa}$

$$\tau = 15.5 \cdot \text{MPa}$$

$$\tau < 0.8 S_{\text{lining}} = 1$$

Screw verification :

Load by screw : $F_{fv} = 875.5 \cdot N$

Filet shear surface in 1 screw : $S_{f_{filet}} := \pi \cdot d_{filet} \cdot \frac{L_{filet}}{2}$ $S_{f_{filet}} = 106 \cdot mm^2$

with : Diameter of thread pitch $d_{filet} = 9 \cdot mm$

Length screwed in the panel $L_{filet} = 7.5 \cdot mm$

Actual shear stress in filets : $Sh_{filet} := \frac{F_{fv}}{S_{f_{filet}}}$ $Sh_{filet} = 8.3 \cdot MPa$

Sh_{filet} value shall be smaller than $0.8 \times S_{screw}$ and $0.8 \times S_{panel}$ according to UG-23(f)

Allowable stress value of panel : $S_{panel} = 118 \cdot MPa$ $Sh_{filet} < 0.8 \cdot S_{panel} = 1$

(1=true, 0=false)

Allowable stress value of screw : $S_{screw} = 92.3 \cdot MPa$ $Sh_{filet} < 0.8 \cdot S_{screw} = 1$